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DESIGN OF A COOLING MACHINE OPERATING WITH CO_2 IN SUBCRITICAL CYCLE

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Abstract. *The purpose of this work is to design an ar-water cooling machine using components that operate with CO_2 (R744) in a subcritical cycle. The machine has a concentric tube condenser option (coaxial) with counter-current flow and a cross-flow finned tube evaporator option. It was also used an alternative semi-hermetic compressor and an electronic expansion valve. The components were designed and selected based on mass and energy equations. The specialized literature was consulted for the acquisition of mathematical models more suitable for the project of heat exchangers, which were programmed in the MATLAB® (MATrix LABoratory) software, assisted by the EES® (Engineering Equation Solver) software. The refrigeration capacity of the evaporator was 3.47 kW, operating at a pressure of 19.7 bar and at an evaporation temperature of - 20 °C. The heat exchange rate between the fluids in the condenser was 4.82 kW, working at a pressure of 45.02 bar and at a condensing temperature of 10 °C. The projected machine has a cycle COP (Coefficient of Performance) of 3.47, which can be used by a research laboratory and will allow the production of cold air at 8.1°C, which can be used to aid in the air conditioning of laboratory rooms and can also be applied in laboratory tests of domestic refrigeration machines.*

Keywords: Refrigeration machine, cooling machine, project, CO_2 (R744), subcritical cycle

1. INTRODUCTION

The emission of some gases used in refrigeration has caused a disturbing environmental impact, as it provides the disintegration of the ozone layer that protects the planet from ultraviolet solar radiation. This situation characterizes a recurrent need to use less harmful gases with the minimum efficiency necessary to guarantee their insertion in the market (Faria, 2013).

CO_2 - Carbon dioxide (R744) is a natural fluid that exists in abundance in nature. It has been the object of research in recent years because it can be a refrigerating fluid that, in addition to not being harmful to the environment, presents high variations in enthalpy and specific mass, when compared to the fluids most currently used (Oliveira, 2013).

There is a lot of equipment based on conventional refrigerant fluids in the market, such as CFCs (chlorofluorocarbonates) and HCFCs (hydrofluorocarbonates). In the meantime, the development of CO_2-fuelled refrigeration machines that can contribute to current research is challenging, it introduces a rather promising and innovative idea within the refrigeration industry (Lorentzen, 1994).

Lorentzen (1993) points out that CO_2 is naturally present everywhere on our planet. The atmosphere contains almost 3000 million tons of this gas, about 0.35 ppm, and several hundred billion tons circulate annually in the biosphere. Although it is a greenhouse gas, its negative impact is minimal when compared to halocarbons and will therefore be reallocated from numerous activities that normally release it into the environment, that is, its discharge into the atmosphere will be avoided.

It is worth mentioning the use of CO_2 as a refrigerant in industrial refrigeration (Cleto, 2008), automotive and residential air conditioning projects and industrial drying projects (Kim et al., 2004).

There is little information in Brazil on manufacturers of CO_2-fuelled refrigeration system components and heat pumps. The CO_2 model is an important alternative because it is a natural, non-harmful fluid for the ozone layer and has some thermal properties, such as the latent heat of vaporization, which makes its application in cooling possible (Silva, 2016).

The high working pressure of CO_2 demands the installation project and safety measures to be done with special criteria, demanding a greater specialization of all the factors and components involved in the system, from the project, to the execution, installation, operation and maintenance (Pereira and Primo, 2012). As carbon dioxide is denser than air, it always flows toward the ground (floor) in the event of a leak. For this reason, it is important to have CO_2 detectors placed in strategic locations together with an exhaust system to renew the air in the engine room air every 10 minutes in emergency situations (Silva, 2008).

For Cleto (2008), one of the greatest advantages of using CO_2 in refrigeration is the reduction of use of fluids that present bigger restrictions (ammonia and hydrocarbons). It is essential for the system with CO_2 to have the same or better energy efficiency level as a conventional system for the same application.

The aim of this work is to analyze the complete project of a refrigerating machine that operates with CO_2 in a subcritical cycle.

2. METHODOLOGY

In order to analyze CO_2 as a refrigerant fluid, the project of an ar-water refrigerating machine using components that operate in a sub-critical cycle was proposed, allowing the development of research that can contribute scientifically to the R744 elevation. This will also allow the study of thermodynamic cycles, the analysis of better choices for evaporation, condensation temperatures to optimize the COP (Coefficient of Performance), analysis of thermal exchange coefficients in heat exchangers, and influence of the conditions of hot and cold sources on the operating cycle.

The projected machine will be installed in a research laboratory and will allow the production of cold air (through the evaporator) that can be used to help a system of air conditioning of laboratory rooms and can also be applied in laboratory tests of domestic refrigeration machines.

This section presents the machine's work cycle and the idealization of each of the four components that make up the refrigeration system, namely: compressor, condenser, evaporator and expansion device.

The remaining accessories necessary for the proper functioning of the system will also be presented.

2.1 CO_2 Sub-critical cycle

A refrigerating cycle diagram is capable of schematically demonstrating the entire system, as well as representing the behavior of the refrigerant throughout the system.

The subcritical cycle is the most widespread commercially when it comes to the use of CO_2 as a cooling fluid and it is showed in Fig. 1. According to Zhang et al. (2011), practical values for condensation and evaporation temperature in a CO_2-based cooling system would be 20°C and -20°C respectively. The compression process is assumed adiabatic, because there is no heat exchange with the external environment. With regard to the condensing and evaporating pressures, these are, in general, around 40 bar and 25 bar, respectively (Silva, 2008).

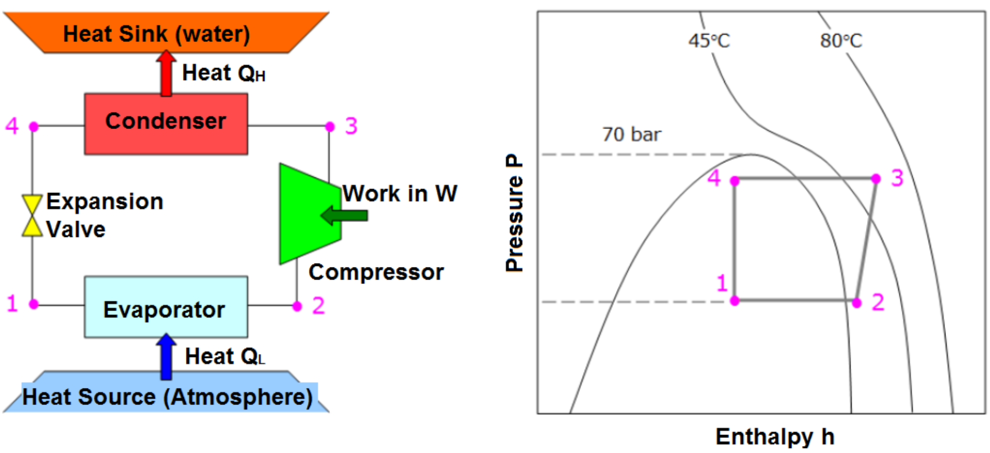


Figure 1. CO_2 Sub-critical cycle. Source: Oliveira (2013).

2.2 Definition of the compressor

The theoretical power of the compressor ($\dot{W}_{comp}$) is the amount of energy in the time unit that must be supplied to the refrigerant by the compressor so that it passes from state 1, in the compressor suction, to state 2, in the compressor discharge, being this an adiabatic process. Considering the scheme of this process shown in Fig. 2 and that there is no heat through the control volume, a mass and energy balance can be presented in Eq. 1 and 2.

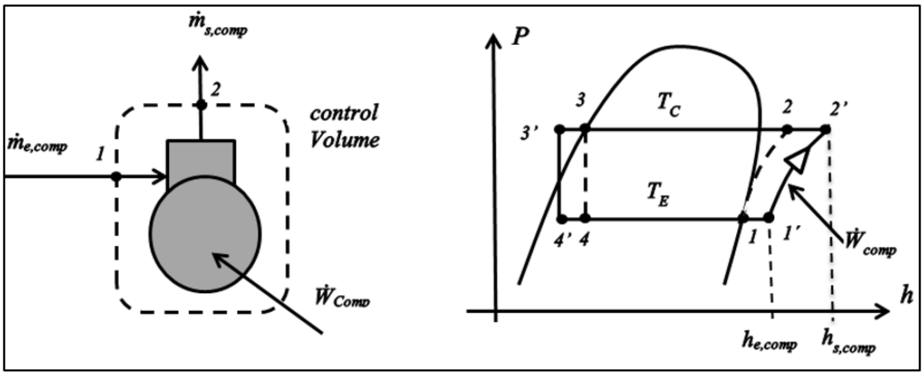


Figure 2. Diagram of the adiabatic compression process in the compressor.

$$\dot{m}_{in,comp} = \dot{m}_{out,comp} = \dot{m}_{comp} \tag{1}$$

$$\dot{W}_{comp} = \left(h_{out,comp} - h_{in,comp}\right)\dot{m}_{comp} \tag{2}$$

Where $\dot{m}_{in,comp}$ and $\dot{m}_{out,comp}$ are the mass flow rates at the compressor inlet and outlet, respectively. $h_{out,comp}$ and $h_{in,comp}$ are the enthalpies at the compressor output and inlet, respectively. Due to the irreversibility of the compression process, there will be an increase in specific entropy between the compressor inlet and outlet, i.e. the real process is considered polytrophic. This effect can be measured by the compression efficiency η_c. The compression efficiency is the representation of the deviation of the real cycle from the ideal, caused by losses during the process. The reasons for these losses include friction and the refrigerant pipe flow losses. Given this situation, η_c can be calculated by Eq. 3.

$$\eta_c = \frac{W_{comp,is}}{W_{comp,re}} \tag{3}$$

Where $W_{comp,is}$ is the isentropic compression work and $W_{comp,re}$ the real compression work.

The mass flow can be determined by using the volumetric efficiency of the compressor, η_{vol} via Eq. 4. For this, the displacement rate t_{des} is calculated and, finally, the mass flow $\dot{m}_{comp}$ can be obtained by means of Eq. 5 and 6.

$$\eta_{vol} = \frac{V_{vol}}{t_{des}} \tag{4}$$

$$t_{des} = n_{cil}\left(\frac{rpm}{60}\right) c \left(\frac{\pi D^2}{4}\right) \tag{5}$$

$$\dot{m}_{comp} = t_{des}\, \eta_{vol}\, \rho_{e,comp} \tag{6}$$

Where n_{cil} is the compressor cylinders number, rpm is the rotation per minute of the motor, c and D are length and cylinder diameter in meters, respectively, and $\rho_{e,comp}$ is the specific weight of the refrigerant in the entrance of the compressor.

The volumetric flow rate at the compressor inlet (V_{vol}) is an initial data. The displacement rate of the compressor is the volume "swept" by the pistons during its path. The valves are operated by springs, so that when the pressure in the cylinder becomes smaller than the pressure of the suction line the suction valve opens, releasing the gas inlet in the cylinder. On the other hand, when the pressure inside the cylinder exceeds the pressure of the discharge line the discharge valve opens, releasing the outlet of the compressed gas from the cylinder.

2.3 Evaporator design project

The evaporator is the refrigerating machine component in which the refrigerant fluid undergoes a change of state from the liquid phase to the gas phase. Being the most important component of a refrigeration system, since it is responsible for the removal of heat from the environment, the efficiency of the system depends directly on the design and proper operation of the evaporator.

According to Zhang et al. (2011) overheating temperatures between 0 ° C and 20 ° C are common in CO_2 refrigeration machines operating in subcritical cycles and are important in preserving compressor life. There is a variation of the COP with the increase of the overheating temperature. Regarding this situation, an overheating temperature of 10 ° C will be considered in this project. There will be a reduction in the COP, however this measure is fundamental to avoid the entry of liquid in the compressor.

In order to improve the heat exchange between the fluids, the evaporator chosen for this project is a coil type evaporator with finned tubes and cross flow. The evaporator chosen can be seen in Figure 3, where heat exchange occurs on the fins delimiting the airflow and the tubes, together with the fins, allowing the thermal exchange between them.

According to Stoecker and Jones (1985), when the temperature of an evaporator, which cools air, drops below 0 ° C, an ice surface will be formed. Ice is damaging once it behaves as an insulator when it forms thicker layers, preventing thermal exchange, and reducing airflow in forced convection coils.

In order to solve this problem, Althouse et al. (2003) proposes the implementation of a bypass system using an electronically operated solenoid valve. When the valve is open, the refrigerant causes defrost in the evaporator due to the heat exchange with the hot gas leaving the compressor. For the operation of the valve, a timer is used to release the electric current that causes it to open, thereby allowing the hot gas to pass through. At the end of the time, the current is cut off and the valve closes, ending the defrosting.

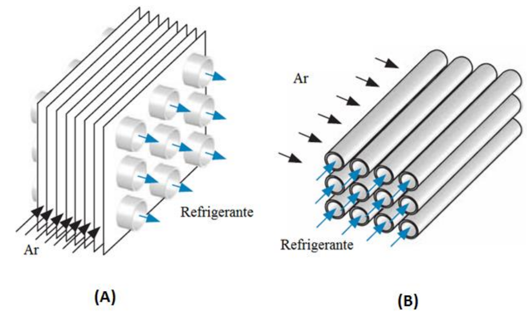


Figure 3. Heat exchanger, (A) finned with cross-flow, (B) without fins.
Source: Nellis and Klein (2009).

For the forced convection it will be necessary to use an axial type ventilator. Once the ability to draw heat from the environment by the crossflow evaporator depends directly on the air velocity produced in the turbomachine, handling that velocity is of utmost importance. For this, it becomes necessary to use an electronic device, the frequency inverter, which is the tool for induction motor driven systems that require variable rotation.

According to Stoecker and Jones (1985), the refrigerating effect is the heat exchanged in the process 1'-4 ', or $h_{out,evap} - h_{in,evap}$ as shown in Fig. 4.

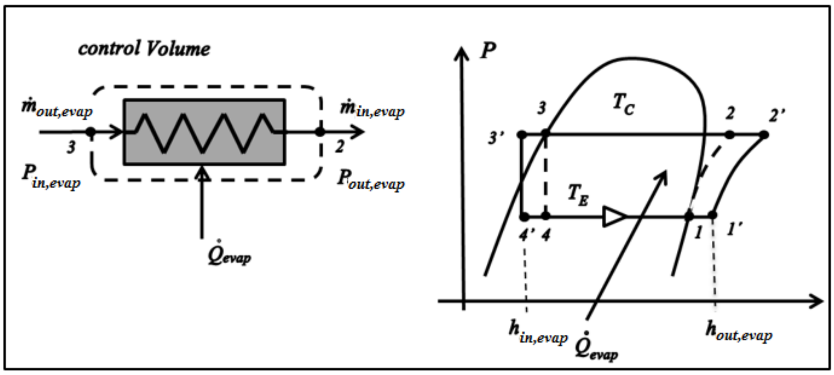


Figure 4. Isobaric evaporation process schema in the evaporator.

Considering the system operating in a steady state, under the project ideal conditions, the energy balance in the evaporator given by Eq. 7 to 9. $\dot{Q}_{evap}$ is the cooling capacity of the cycle operating at the temperatures T_C, condensation, and T_E, evaporation.

$$\dot{m}_{in,evap} = \dot{m}_{out,evap} = \dot{m}_{evap} \quad (7)$$

$$P_{in,evap} = P_{out,evap} \quad (8)$$

$$\dot{Q}_{evap} = \dot{m}_{evap}(h_{out,evap} - h_{in,evap}) \quad (9)$$

According to Stoecker and Jones (1985) when the temperature of an evaporator, which cools air, drops below 0 ° C, surface icing will occur. Ice is damaging to behave as an insulator when it forms thicker layers, preventing thermal exchange, and reducing airflow in forced convection coils. To solve this problem, Althouse et al. (2003) proposes the implementation of a bypass system using an electronically operated solenoid valve. When the valve is open, the refrigerant causes defrost in the evaporator due to heat exchange with the hot gas exiting the compressor. For the operation of the valve, a timer is used to release the electric current that causes the valve to open, consequently allowing the hot gas to pass through. At the end of the count the current is cut off and the valve closes, ending the defrost.

For the forced convection it will be necessary to use a axial fan type. Since the ability to draw heat from the environment by the cross flow evaporator depends directly on the speed of air produced in the turbomachine, handling that velocity is extremely important. For this, it becomes necessary to use an electronic device, the frequency inverter, which is the support of induction motor driven systems that require variable rotation.

2.4 Condenser design project

The condenser used was designed according to the model proposed by Yamagushi et al. (2011). The water flows upwardly between the tubes and the refrigerant flows downwardly into the countercurrent inner tube. Yamaguchi et al. (2011) developed a simulation model for CO_2 heat pump. In the research it was necessary to develop a coaxial heat exchanger to act as a cooler (in the case of CO_2 the term condenser is changed for cooler once this fluid does not change phase, remaining steam during the heat exchange process). Figure 5 shows the schema of the coaxial heat exchanger mentioned above. Its constructive aspects are identical to the one used in this work.

In a heat exchanger, fluids flowing at different temperatures provides heat exchange between them. The refrigerant heat loss from necessary to perform its condensation, subcooling and de-superheating can be easily calculated using Eq. 10 to 12, where $\dot{m}_f$ is the mass flow rate of refrigerant and $h_{in,cond}$ and $h_{out,cond}$ are the input and output enthalpies, respectively. The scheme of the condensation process represented on a simplified P-h graphic can be seen in Fig. 6.

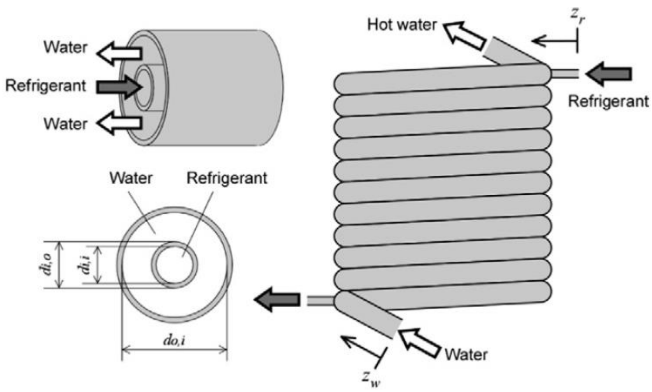


Figure 5. Idealized coaxial capacitor.
Source: Yamaguchi et al. (2011).

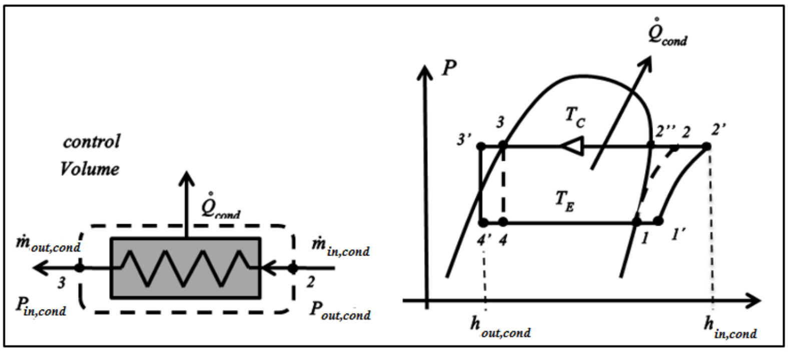


Figure 6. Isothermal heat rejection in the condenser.

$$\dot{m}_{out,cond} = \dot{m}_{in,cond} = \dot{m}_f \quad (10)$$

$$P_{in,cond} = P_{out,cond} \quad (11)$$

$$\dot{Q}_{cond} = \dot{m}_f (h_{out,cond} - h_{in,cond}) \quad (12)$$

The heat exchange section length is increased so that the refrigerant provides de-superheating (2 to 2"), undergoes condensation (2" to 3), and even subcooling (3 to 3') before entering the expansion device.

2.5 Expansion device

According to Stoecker and Jones (1985), the expansion device has two purposes: to reduce the pressure of the refrigerant while liquid and to regulate the flow of refrigerant entering the evaporator. The device is considered isenthalpic, once no heat exchange occurs with the external environment. The diagram representing the expansion valve and the simplified graph P-h can be seen in Fig.7. Eq. 13 and 14 represent energy and mass balances by the device.

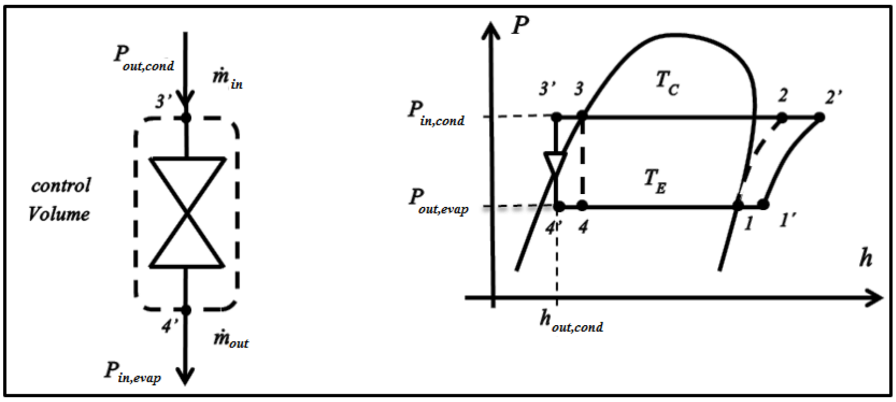


Figure 7. Adiabatic expansion process in the expansion valve.

$$\dot{m}_{in} = \dot{m}_{out} \quad (13)$$

$$h_{out,cond} = h_{3'} = h_{4'} \quad (14)$$

According to Faria (2013), the control of the refrigerant flow that enters the evaporator is usually realized using capillary tube, thermostatic expansion valve or electronic expansion valve. The use of the electronic expansion valve for the control and optimization of heat transfer in the evaporator results in superior performance characteristics when compared to traditional devices such as capillary tube and thermostatic valve, mainly operating with transient and non-linear conditions. According to Silva (2008), CO_2 as a cryogenic fluid requires a very fast response time during its expansion in low temperature evaporators to enable freezing, so the electronic valves are those that meet this requirement.

2.6 Coeficient of performace

According to Stoecker and Jones (1985) the performance coefficient, also known as efficiency coefficient, defines the performance of a refrigeration cycle. The COP is the ratio of the amount of what is desired to the amount of what is spent. In Fig. 8, a standard cycle can be seen and the equation to obtain the coefficient of performance is given by Eq. 15.

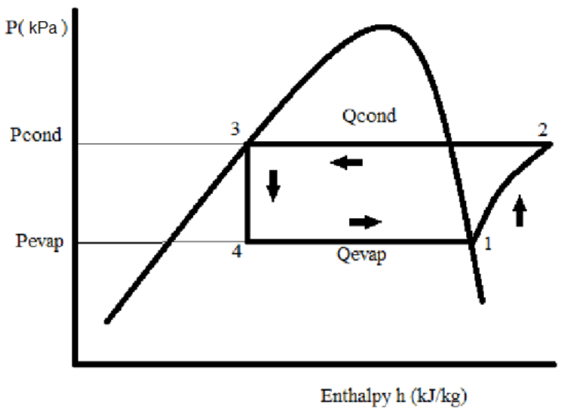


Figure 8. Vapor compression cycle Pressure-enthalpy diagram.

$$\mathrm{COP} = \frac{h_1 - h_4}{h_2 - h_1} \tag{15}$$

2.7 Other components

This section presents the other components necessary for the proper functioning of the refrigeration system.

2.7.1 Dryer filter

According to Althouse et al. (2003) the refrigeration machine efficient depends directly on the purity of the refrigerant. All impurities such as dust and water should be removed. As contaminants often detach from the components and circulate through the system charged with the refrigerant fluid, a component is needed to make a filtration. This device is the dryer filter. The conventional dryer consists of a cylinder, made of brass, copper or steel, filled with activated alumina or silica gel. These chemicals can absorb 12 to 16% of their weight in water. This component is installed in the liquid line, preferably before the expansion device.

2.7.2 Oil separator filter

According to Althouse et al. (2003) an oil separator collects the lubricant contained in the refrigerant. The oil is collected up to a certain level, being fowerded to the compressor after this level has been reached. This device is located before the condenser and after discharge of the compressor.

2.7.3 Low pressure liquid acumulator

According to Althouse et al. (2003) compressors are damaged when the refrigerant in its liquid state crosses the suction line. In this location, the fluid must already be completely vaporous. In addition to providing an increase in the temperature of the fluid after evaporation (overheating), a liquid accumulator, also known as an anti-shock bottle, is normally used just before the compressor’s inlet. As a result, the liquid droplets are deposited in this bottle and are prevented from crossing into the compressor. This component is located on the low pressure side of the cycle.

2.7.4 Separador de líquido em alta pressão

According to Althouse et al. (2003) the liquid separator is a refrigerant fluid storage tank. The refrigerant fluid is pumped from several parts and stored in the liquid receptor during maintenance. Usually a liquid separator is located after the condenser. This location is not chosen randomly. It prevents bubbles of uncondensed vapor from reaching the expansion valve, which compromises its efficiency.

3. RESULTS AND DISCUSSION

The design of the idealized machine and the components required for its operation are shown in Figure 9.

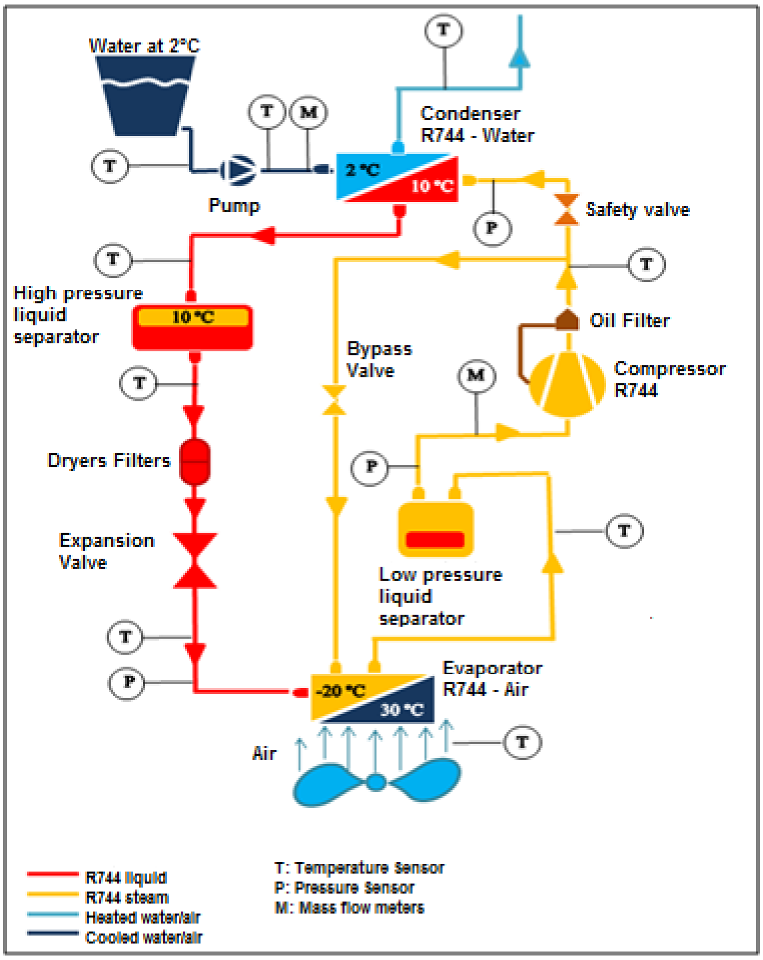


Figure 9. Idealized refrigerating machine and components.

3.1 Compressor selection

In general, the machine under analysis was designed to use a reciprocating semi-hermetic compressor of 1.08 kW power based on pre-set condensation and evaporation temperatures of respectively 10 ° C and -20 ° C.

The semi-hermetic reciprocating compressor has an efficient motor, a suction gas cooler, excellent valve plates, wear-resistant drive mechanisms and variable power regulation. For this project, the smaller capacity compressor was chosen. An operating point was selected for this compressor according to the evaporation and condensation temperatures already mentioned.

From the defined operating point, a software provided by the manufacturer was used for calculations of compressors. Using the input information according to the working conditions, the calculations are done automatically. The results obtained were recorded in Fig. 10 and 11.

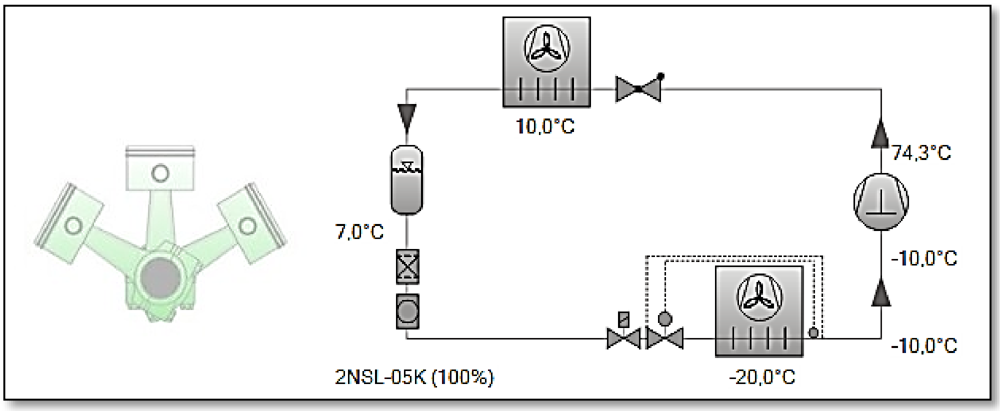
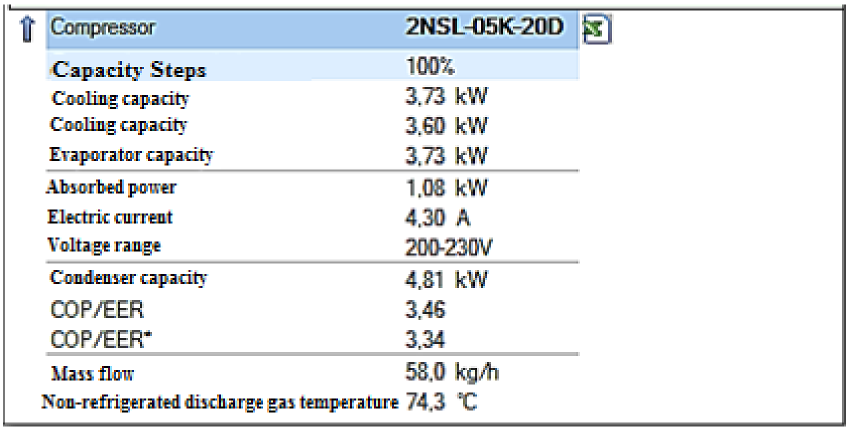


Figure 10. Cycle calculations results using the manufacturer's software.



Compressor	2NSL-05K-20D
Capacity Steps	100%
Cooling capacity	3,73 kW
Cooling capacity	3,60 kW
Evaporator capacity	3,73 kW
Absorbed power	1,08 kW
Electric current	4,30 A
Voltage range	200-230V
Condenser capacity	4,81 kW
COP/EER	3,46
COP/EER*	3,34
Mass flow	58,0 kg/h
Non-refrigerated discharge gas temperature	74,3 °C

Figure 11. Calculations results of the other system parameters using the manufacturer's software.

3.2 Evaporator design project

The mathematical modeling of the project was programmed in the MATLAB® software (MATrix LABoratory) and the input data sheet for calculations was developed in the MICROSOFT OFFICE EXCEL® software. Properties for R744 were taken from the CoolProp package for MATLAB® containing the thermodynamic properties of various fluids. A graphical interface for the results of the evaporator design has been developed in MATLAB®, providing a pleasant interface and easy manipulation. The library of functions and properties of the EES® (Engineering Equation Solver) was also used to perform some calculations.

For the evaporator design project it was necessary to separate the refrigerant flow in two regions: evaporation and overheating. The determination of the heat exchange convective coefficients from the refrigerant (single phase and biphasic) and from the secondary fluid (air) were performed using the correlations present in the literature, and the responses were compared for the biphasic region (CO_2), defining the most appropriate to the project. In the region of single-phase CO_2 flow, the correlations of Dittus-Boelter, Sieder and Tate, and Gnielinski were used. The correlations developed by Kim, Youg and Webb were used for the air flow. In the biphasic flow region (CO_2), the Kandlikar and Jung correlations were used. The complete heat exchanger mathematical modeling is available in Silva et al. (2017b). Fig. 12 shows the window with the output variables and graphs of Kandlikar and Jung convective coefficients. The red message alerts if the heat rate taken by the proposed heat exchanger is greater than the required heat rate.

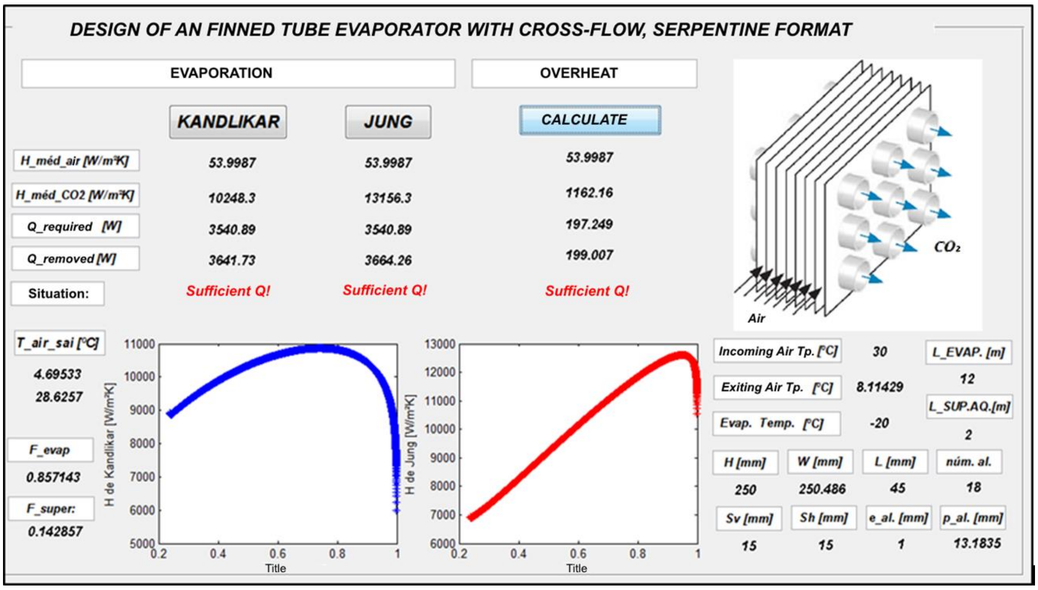


Figure 12. Evaporator project design results.

The cooling capacity of the evaporator was 3.74 kW, operating at a pressure of 19.7 bar and the evaporation temperature of - 20 ° C. Once the geometry was defined, as results can be highlighted the air temperature at the evaporator outlet, 8.1 ° C, and the total length of the refrigerant tube, 14 meters. The chosen exchanger heat is quite compact, with a height and width of approximately 0.25 meters, depth of 0.045 meters and has only 18 fins. After the modeling was completed, a drawing of the evaporator was made in SolidWorks®, which can be seen in Fig. 13.

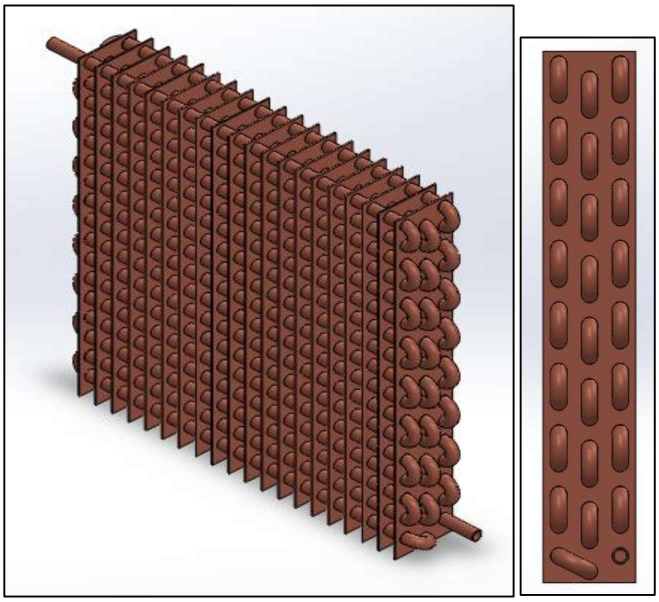


Figura13. Evaporator model design.

3.3 Condenser design project

The mathematical modeling of the project was programmed in MATLAB® software and an input data sheet was created for calculations using EXCEL® software. Properties for R744 were taken from the CoolProp package for MATLAB®.

For the design of the condenser it was necessary to separate the refrigerant flow into the condenser in three regions: desuperheating, condensation and subcooling. The determination of the heat exchange convective coefficients from the refrigerant (single phase and biphasic) and secondary fluid (single phase) were performed using the correlations present in the literature, and the responses were compared for the biphasic region (CO_2) and defined the most appropriate to the project. In the single phase flow region, the correlations of Dittus-Boelter, Sieder and Tate, and Gnielinski were used. In the region of biphasic flow (CO_2), the correlations of Akers, Shah and Chato were used. CO_2 loss was also determined in all regions, by the Yoon equation for the single-phase region and by the Muller equation in the biphasic region. The complete heat exchanger mathematical modeling is available in Silva et al. (2017a). With a didactic purpose, a graphical interface was created from MATLAB® containing buttons for calculations of the length, according to the desired correlation, load loss of both water and refrigerant and the water exit temperature each section according to Fig. 14.

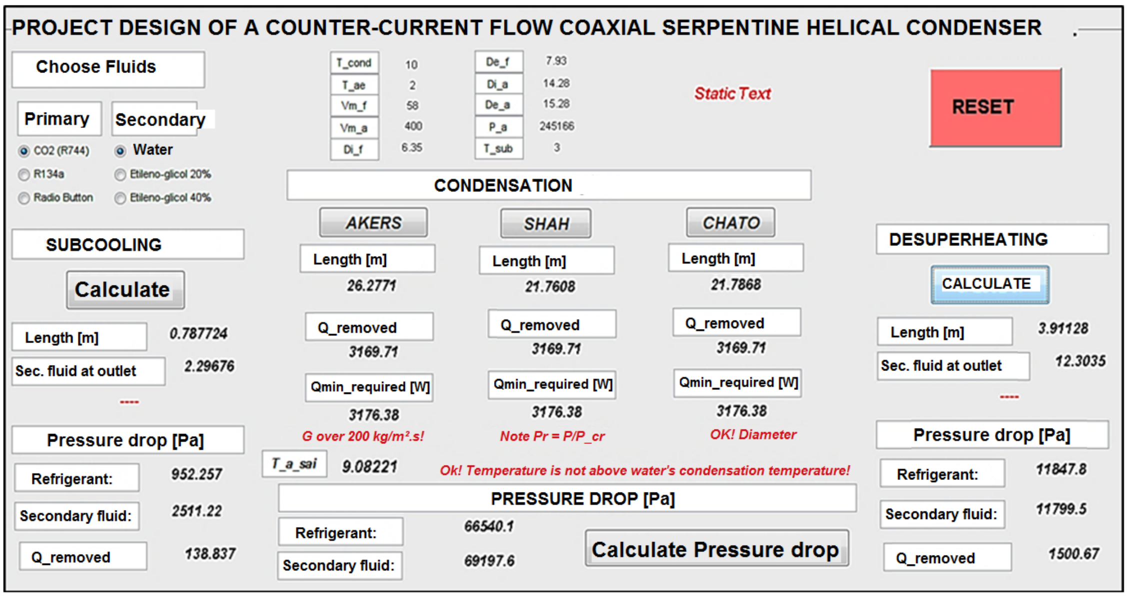


Figure 14. Condenser project design results.

A final value of 26.5 meters was obtained for the coaxial condenser length in counter current flow. The refrigerant pressure loss was calculated to analyze how much the actual situation differs from the ideal situation. In the condensation section, the total pressure loss was approximately 58 kPa, and this pressure variation can not be detected by the manometer sensitivity. On the other hand, the total pressure loss in the subcooling, condensation and de-superheating sections for the secondary fluid was obtained for the water pump sizing. The lengths of 1 m were considered for the water to travel before and after thermal exchange with the refrigerant. For the secondary fluid, the approximate value of 74 kPa of pressure loss was found.

After completing the modeling, a condenser drawing was made using SolidWorks®, the drawing can be seen in Fig. 15. This exchanger needs to be surrounded by an insulating material so that the thermal exchange takes place without external interference.

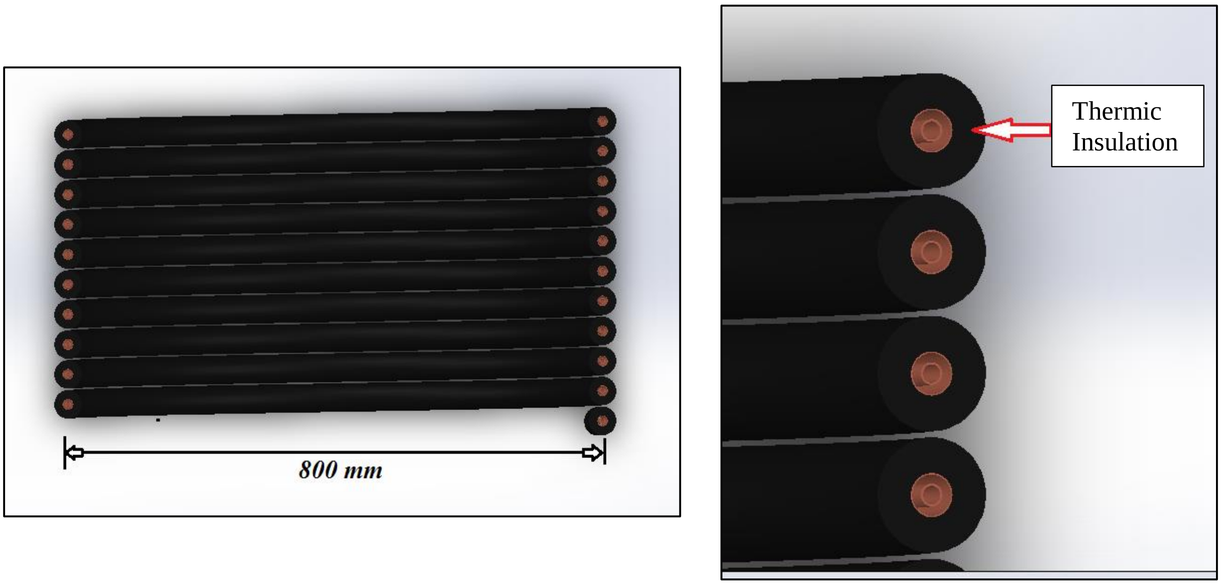


Figure 15. Condenser model design.

The heat rejection rate from the refrigerant in the condenser was 4.82 kW, operating at a pressure of 45.02 bar and the condensation temperature of 10 ° C.

4. CONCLUSIONS

In general, the machine under analysis was designed to use a reciprocating semi-hermetic compressor of 1.08 kW power based on pre-set condensation and evaporation temperatures of respectively 10 ° C and -20 ° C. The condenser will be coaxial flow countercurrent tubes type with a heat exchange rate of 4.82 kW. The evaporator will be a cross-flow finned tubes type with refrigerating capacity of 3.74 kW. The expansion device will be an electronic valve.

The defined cycle COP takes in consideration the input and output enthalpies of the compressor as well as the input of the evaporator. The theoretical cycle COP was 3.47. The air temperature in the evaporator outlet was 8.1 ° C, and the air can be piped and distributed to a room air conditioning system and make it available for the domestic refrigeration machines research laboratory.

The purpose after the assembly of the machine is to contribute with the current research in refrigeration, regarding the need to eliminate the gases that have chlorine in its composition, through experiments. From the beginning it was highlighted the fact that CO_2 presents itself as an important alternative because it is a natural fluid and not aggressive to the ozone layer, besides possessing thermal properties that makes possible its application in refrigeration.

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6. RESPONSIBILITY NOTICE

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